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EDITORIAL

Railway lubrication occupies a position of great importance in the field of lubrication. While journal lubrication constitutes the largest single item, it is in the lubrication of gears and pinions that the greatest difficulties are encountered, and that the greatest wear and tear takes place because of defective lubrication. The articles in this issue constitute a complete treatise on the subject. Crater Compound is the ideal lubricant for this class of work.

In view of the prejudice which still exists in some circles against lubricating oils made from asphaltic base crudes, a statement in the February, 1917, issue of The Petroleum World, published in London, is interesting. In an article entitled "German Drilling in Roumania," written shortly after the Germans gained control of the valuable Roumanian oil fields, the question is raised as to the value of these fields to Germany. Because of the wrecking of the oil producing wells and the destruction of derricks, machinery, refineries, storage tanks and pipe lines, the existing facilities were rendered useless, and it was a question of how long it would take the Germans to drill new wells. "In some of the captured fields," says The Petroleum World, "drilling is both quick and easy. A couple or three months suffices to get a small producer in certain selected areas, but the oil thus got is of paraffine base, and not as valuable to Germany as some other Roumanian oil of asphaltic base, much better for the manufacture of lubricating oils, which are Germany's chief requirement. (We have stated this repeatedly in The Petroleum World, and evidence from Roumania confirms it.) Germany would, then, have to decide whether to devote the initial energy of her new drilling to quick output of paraffinous oil or to wells in districts that give crudes affording lubricating oils of high quality, but that take somewhat longer to drill and bring in."

OPERATING CONDITIONS OF RAILWAY MOTOR GEARS AND PINIONS*

By A. A. Ross†

THE limitations imposed upon the space available for gears and pinions in the design of modern railway motors, and the severe conditions under which railway motor gears and pinions are operated, necessitate the use of materials which will insure protection against breakage and secure the maximum resistance to abrasive wear. While the gear and pinion manufacturers have been striving to meet these conditions with various grades of steel and special methods of treating the steel, very few operators have taken steps to improve operating conditions.

Apparently the average operator has never given this side of the question serious consideration. A pinion is a pinion and a gear is a gear, and if they break or wear out rapidly it is simply defective material and up to the manufacturer to make good. If the manufacturer does not feel so inclined the operator invariably changes to some other manufacturer's product, and in nine cases out of ten the original trouble recurs. Had the operating conditions been definitely known, the trouble might have been overcome with the original material and the operator, the manufacturer, and the trade benefited thereby. . . .

Variation in Gear and Pinion Life. . . . From actual service observations it is evident that in straight carbon steel or non-alloy steel the harder the wearing surface of the tooth, the greater the resistance to abrasive wear. Case-hardened material now offered to the trade under various trade names by practically all gear and pinion manufacturers affords about the hardest possible tooth surface, but it does not afford maximum protection against breakage. While not wishing to offer excuses for manufacturing defects, the structure of this steel with its glass-hard brittle surface is very susceptible to injury from shocks such as may be transmitted to the teeth during motor flashovers or when at high speed the wheels hit high rail joints, frogs, etc., but the greatest source of danger lies in the operator's methods of mounting and dismounting; this will be referred to later. The material is also very expensive to manufacture, but its total life has been so much greater than the old combination, which consisted of oil-treated pinions and untreated cast steel gears, that its first cost and breakage have been overlooked by the operator.

With the advent of the less expensive specially treated homogeneous steel having physical characteristics which afford maximum protection against breakage and a high uniform hardness which resists abrasive wear almost to the same degree as case-hardened material, it devolves upon the operator to determine by actual service tests on his own equipment which is the most profitable.

It is unsafe for one operator to use the life values established on

* This article, in more complete form, was orginally published in the *General Electric Review*, Vol. XVIII, No. 4, April, 1915, pp. 249-258. Revisions by the author have been incorporated into the present article.

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some other road, for the life is in a great measure affected by the ratio. Still it is quite common to find a radical variation in the life of the same grades of gears and pinions on two roads which have duplicate equipments. Such variations can only be accounted for as follows:

The first and greatest factor is the grit or cutting substance which accumulates in the gear pan and, when mixed with the lubricant, acts as an abrasive lap on the gear, pinion, or both. Practically every master mechanic will disclaim its presence, but it is there, and it has been found in quantities as high as twenty-four per cent.

It is doubtful if the reader appreciates what even one per cent of sand really means. The average amount of lubricant in gear pans is about twelve pounds. The quantity of sand in the vial (Fig. 1) represents one per cent of sand in twelve pounds of lubricant. The vial is one and one-half inches in diameter. Pause for an instant and consider what ten per cent or ten times, or twenty-four per cent or twenty-four times, this quantity means.

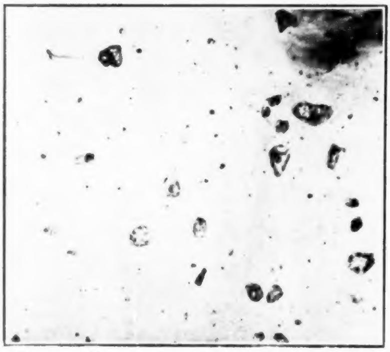
Fig. 2 shows photo-micrographs of a popular motor grease containing various percentages of sand magnified to 26 diameters. The sand used in the mixture was first screened through an 80-mesh screen.

The sand usually enters between the gear hub and gear pan in the form of street dust, brake shoe dust, and wheel wash. Very often the contour of the web and hub of the wheel is such that the natural flow of the wheel wash is against the opening in the gear pan, as shown in Fig. 3. It would be much better if this were retarded by either making the diameter of the wheel hub larger or smaller than the gear hub. The clearance between the gear hub and the gear

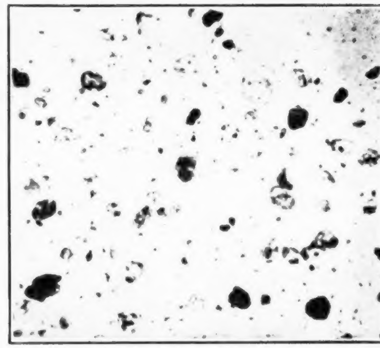
Figure 1
Sand shown in vial represents one per cent by weight in 12 pounds of gear lubricant

pan is sometimes enclosed by a felt dust guard, but this is not a permanent protection as the felt soon fills up with sand and breaks off. Carelessness when adding lubricant or when the lower half of the pan is lying in the pit during inspection is another source of dirt getting into the grease. One of the largest operators in the East traced the cause of rapid wear to the presence of sand in the lubricant which was carried into the pan by the wheel wash during a period of heavy snow and slush. The sand or stone dust seems to scour the lubricant off the teeth.

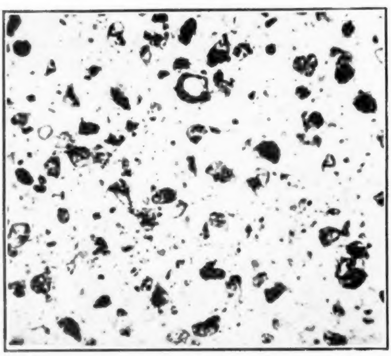
It has recently been proved from actual tests that the distance between the tops of the gear teeth and the bottom of the pan has a great effect on the damage which the grit in the grease will do to the gearing. A test was made on two equipments; one with about $1\frac{1}{2}$ inch clearance and the other with $\frac{1}{4}$ inch clearance between the gear and bottom (inside) of the pan. The grease in both pans showed about the same amount of grit. The life of the pinions in the pan with $\frac{1}{4}$ inch clearance was only about half of those in the pan with



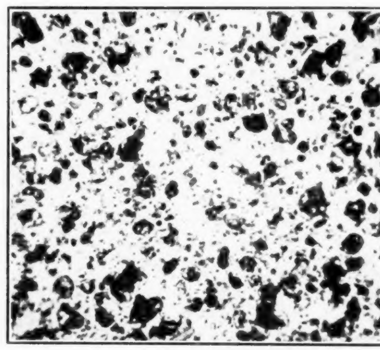
Grease Containing 2 Per Cent of Sand by Weight



Grease Containing 5 Per Cent of Sand by Weight



Grease Containing 10 Per Cent of Sand by Weight



Grease Containing 25 Per Cent of Sand by Weight

Figure 2

Photo-micrographs (26 diameters) of popular motor grease containing various percentages of sand

1½ inch clearance. In other words, there was sufficient space in the pan with the 1½ inch clearance for the grit to settle to the bottom and not get in contact with the gear teeth, whereas in the pan with ¼ inch clearance the grit was continually being churned up with the grease.

Before shipment the finished surfaces of the gears and pinions are given a coating of slushing compound to prevent rust. This should be removed before the gears and pinions are placed in service as it becomes filled with lime and sand during shipment.

The second factor is excessive lining wear, which permits improper mesh. The design of the motor and the truck prevents the use of a bearing on each side of the gear and pinion, so that when power is transmitted from the pinion to the gear both the armature and axle shafts tend to spring diagonally away from one another.

The armature shaft bearing adjacent to the pinion becomes worn on the side which is farthest from the axle, while the bearing on the other end wears on the side near the axle. The wear on both bearings is not radial with the center of the axle but at two points about forty-five degrees above and below the radial line, according to the direction of

rotation. This allows the pinion teeth to set at an angle to the gear teeth, which means that the ends of the teeth next the motor receive the greatest percentage of shocks and must perform the major portion of the work. Such conditions account for the greater wear on the motor ends of the teeth. Consequently, very little would be gained by increasing the present standard width of face. Furthermore, the axle linings wear on the upper portion of the bore away from the armature shaft, which tends to carry the pitch lines of the gear and pinion still further apart and forces the working contacts to take place on the top or excessive friction points.

Some operators claim that it is impossible to maintain $\frac{1}{8}$ inch as the limit for axle lining wear, as it forces too frequent renewals, and that the slight reduction in gearing maintenance is offset by an increase in lining maintenance. If gearing maintenance were the only consideration there might be some ground for the argument, but the improper mesh also produces a noisy chattering, which affects commutation and the nerves of the population living along the route over which the gearing is operated. If the operator doubts the effect on the rest of the equipment, let him get into the pit under a motor with the axle linings worn, say $\frac{1}{4}$ of an inch, start the car so that the pinion climbs the gear, and, as the car gains momentum, note the blow on the axle as the motor settles back into place. This is especially severe on heavy equipments. . . .

Many of the latest types of motors are provided with axle dust guards which completely enclose the axle between the bearing housings and prevent sand from getting into the linings. This is a step in the right direction, but there is a

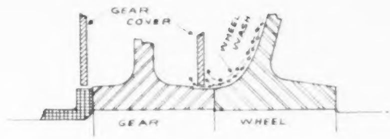


Figure 3
Diagram showing the natural direction of flow of wheel wash against the gear cover

tendency on the part of the operator to pay less attention to axle lining wear limits owing to the trouble of removing the guards for inspection. A very fair idea of lining wear without removing the guard can be obtained by jacking up the bearing housing with a block and pinch bar.

The third cause is the consistency of the lubricant. There are many good grades on the market, but, be the grade what it may, the lubricant should be of such consistency and used in such quantities that the gear teeth will dip frequently. The writer has absolutely no confidence in the virtue of a lubricant so hard that the gear cuts a groove through it and as soon as the grease in the groove is deposited on the sides of the pan, the teeth run dry. The lubricant should be soft enough to level back and be again picked up by the gear teeth. Also, if the lubricant is soft the sand will more readily settle to the bottom. The operator who uses a summer grade during the winter months usually does so at the expense of his gearing.

The number of stops in the schedule, the coasting limits, and the motormen's methods of accelerating and braking may be considered the fourth factor.

Operator's Responsibility for Broken Gears and Pinions.—By comparing the thickness at the base of the gear and pinion teeth, it will be noted that the thickness is less on the pinion. This is due to the undercut at the root of the pinion tooth

to give clearance for the gear tooth. The pinion teeth being the weaker of the two, it is obvious that the greatest percentage of tooth failures will occur on the pinion and, as previously mentioned, on the motor ends of the teeth. The breaks usually begin in a V-shaped fracture and progress irregularly to the top of the tooth about one-third across the tooth face. The usual question is: What causes the failures, and why should one operator have more than another? Invariably the operator will put the responsibility up to the manufacturer, but there are many causes for failures over which the manufacturer has absolutely no control and for which the operator is entirely responsible, especially the methods employed when mounting and dismounting the pinions. The most common and injurious method for mounting is that of driving a pinion home with a sledge. Usually one man holds a babbitt metal ring or cup-shaped protector over the end of the pinion while another swings on to it with a ten or twenty pound sledge.

Now let us consider what happens to the pinion. The shaft on which the pinion is to be mounted has a tapered fit and every blow of the sledge adds internal stresses to the body of the pinion; the maximum stresses depending on the proficiency of the man who swings the sledge and the number of blows delivered. The section of metal between the root of the tooth and the bore at the large end or between the root of the tooth and the bottom of the keyway on the popular pinions, 14 and 15 teeth, 3 pitch; 17 and 18 teeth, 2½ pitch, is usually just about thick enough to take care of the tooth load stresses, and if excessive stresses are added when mounting, the pinion after being placed in service either splits through the keyway or a portion of one or more teeth directly over the keyway fails from fatigue at low mileage.

Some operators make it a practice to heat the pinions in boiling water and then sledge them on as just described. This method is exceptionally severe, for the metal is subjected to shrinkage strains plus the stresses set up by sledging.

The following is the safest method and is giving perfect satisfaction on several large roads: The pinion is first slid on to the shaft to make sure that it fits properly and especially that it does not ride the key. It is then placed in boiling water until it is heated clear through, about forty-five minutes. (Flame or heating in an oven should never be allowed, as the temper of some grades may be easily drawn and the virtue of the treatment lost.) It is then seated quickly on the shaft and rammed home by means of a round hard wooden block about twelve inches long, cupped at one end to clear the shaft, and hooped at the ends to prevent splitting. A handle about two feet long should be inserted. When completed the block is like a wooden maul except that one end is cupped. The maul should be placed against the pinion and struck one blow with a ten pound sledge. The nut should be immediately set up and the block struck once more and the nut again set up. A wooden block as above described cushions the injurious blows of the sledge and distributes the blow evenly on the pinion. A great many operators use a metal babbitted ring as a protection, but this is injurious and does not have the cushioning effect of a wooden block. If the pinion is heated clear through and fits the shaft properly it will not work loose.

The fit on the shaft is very important. If the shaft is not perfectly round or if the metal is swelled on each side of the key, which can easily be done by using force to seat a key which is too wide for the keyway, or if the pinion rides the key, there is a tendency for the mounting stresses to localize at one of the bearing points in the body of the pinion, which eventually results in a tooth failure or a crack in the body directly over the point of localization.

The method of dismounting is another great source of pinion tooth failures. It is still common practice to drive wedges between the bearing housing and the end of the pinion. The writer doubts if it is possible to remove a pinion in this manner without subjecting one or more teeth to injurious shocks, especially so with case-hardened material, which is easily damaged and which is removed several times during its life.

A pinion puller which grips only two or three teeth is equally bad. A puller which grips all the teeth is the safest. There are two or three on the market which can be assembled on all types of split frame motors, providing the armature is removed from the motor, but on some of the old types of box frames it is impossible to assemble them unless the outside diameter of the pinion teeth is at least 3/4 of an inch greater than the projection shown as C diameter in Fig. 4. Space B will permit a puller jaw thick enough to withstand the stresses. On the later type of box frame motors the frame heads are chamfered as shown at D. This plus the space A, which is usually one-quarter of an inch, will accommodate a puller jaw similar to E and permit the use of satisfactory pullers. Rather than use wedges it

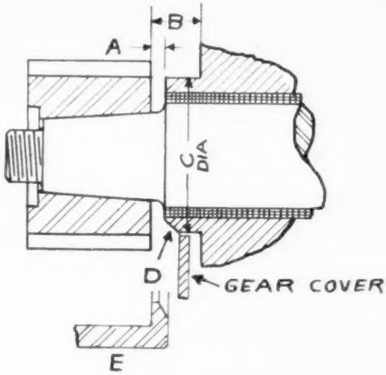


Figure 4

Diagram of Frame Head and Pinion showing clearance for pinion puller ring

would be more economical for the operator to chamfer his frame head in the same manner.

On the other hand, all steel will reach its life limit and break down from what is known as "fatigue" if the load is repeatedly applied, even though the load is far less than it can bear indefinitely. This accounts for a great percentage of tooth failures on heavy equipments where the tooth loads closely approach the elastic limit of the material. The failures may occur before the teeth have even approached the wear scrapping point. In such cases the operator should establish a safe mileage limit and scrap the pinions regardless of the amount of wear.

There are so many different grades of gears and pinions on the market that the master mechanic is sometimes uncertain as to which grade of pinion is the most economical to operate with a certain grade of gear.

In the first place it is unreasonable to expect satisfactory results from a combination in which the gear is harder than the pinion, for each pinion tooth makes from two to four contacts to each contact of a gear tooth. From actual service

observations it is evident that the best results will be obtained by operating together a gear and pinion of the same hardness. This means that the gear will outwear two and sometimes three pinions, and the second and third must mesh against worn gear teeth, which will produce noise until the gear and pinion teeth have adjusted themselves to the proper contour. While this adjustment is taking place the wear seems to have a greater effect on the new pinion. The total life of the second and third pinions will be less than the first or original pinion. Some master mechanics ask: Is it possible for me to obtain a pinion which will give a life equal to the gear? With the grades of gears and pinions on the market at present the writer's answer—from an economical standpoint—is "No." It can, however, be accomplished by operating a hard pinion and soft gear together, but I think the trade learned from experience when the case-hardened pinions first came on the market that such a combination resulted in a reduction in life of the gear, and the gear being the more expensive member of the two, it was anything but a profitable combination.

The operator should, therefore, carefully watch the performance of his gears, for he may be getting high pinion life at the expense of his gears.

There seems to be an extreme and unreasonably wide range in the total tonnage limits which various master mechanics have adopted for mounting solid gears without keys. The range varies from 20 to 80 tons. For city service 20 to 30 tons is sufficient and is proving satisfactory on several roads. For interurban service anything over 50 tons seems unreasonable. For instance, take 40,000 lbs. as equipment weight, 15,000 lbs. for passengers, 30 in. wheel and 5 in. axle and the torque at the gear bore at the slipping point of the wheel is as follows:

40,000 lb. weight of car equipment
15,000 lb. weight of passengers

55,000 lb. total
0.30 per cent coefficient of friction

16,500

$$\frac{16.500}{4 \text{ (no axles)}} = 4{,}125 \text{ lb. per axle to slip at 0.30 per cent coefficient of friction.}$$

$$\frac{30}{5} \times 4{,}125 = 24{,}750 \text{ lb.}$$

$$= 12\tfrac{1}{2} \text{ tons torque at gear bore.}$$

Considering the usual finish on the axle and bore and the ironing effect between the two surfaces when the gear is mounted, it will take a much higher torsional pressure to twist the gear on the axle than the pressure used to mount it. It is common practice to use the same limits for gears and wheels. The high ranges may be necessary for the wheels on account of the end thrusts, but as there is no lateral pressure on the gear the torsional pressure only is to be considered. Of course there are unknown torques to be considered which may result from momentum when motors are short circuited or suddenly reversed. In the above case had the gear been mounted at 20 tons there would have been a sufficient factor of safety to take care of the unknown torque. Therefore why use the high tonnage range, which makes it difficult to remove the gear and subjects the gear to unknown internal stresses which may cause failures, when it can be avoided?

LUBRICATION OF GEARS AND PINIONS ON ELECTRIC RAILWAY MOTORS

By H. K. Eilers
Staff Engineer, The Texas Company

IT is seldom that such severe conditions of gear operation are found as on the cars of electric railways. Even the casual observer can obtain some idea of the enormous stress to which railway gears and pinions are subjected by considering the continual starting and stopping of the cars—the former often accomplished by sudden application of power and the latter by slamming of emergency brake or in some cases by reversing the motor—and also the shocks in passing over switch points, frogs, etc. The result is that these gears quickly wear out, causing not only the expense of renewal but also frequent interruptions of service and consequent dissatisfaction on the part of the operator.

Much systematic and careful study has been devoted to the subject by the various machine manufacturers in order to produce gears and pinions that would withstand the severe operating conditicns, and not wear out or be destroyed too quickly. There has been much improvement along this line, but proper design and manufacture alone will not solve the entire problem. Regardless of how nearly perfect their construction and the material may be, these gears will not give the maximum of service unless they have proper lubrication.

Men in charge of this kind of equipment, as well as manufacturers of motors and gears, have realized the importance of the part played by lubrication, not only in assisting the efficient operation of gears but also in prolonging their life. It has been generally understood for some time what constituted the ideal gear lubricant, although the attempts to produce this ideal lubricant have been few. This is not surprising, perhaps, when we consider the difficulties encountered in producing lubricants that are somewhat out of the ordinary.

Many operators have attempted to meet these severe conditions with so-called gear and pinion greases, gear shields, etc., most of which are only light oils "doped up" with non-lubricating materials or fillers to give them body. Some of these fillers consist of pine tar, resin, talc, etc., all of which are of themselves quite worthless as lubricants and often separate out in use. Such greases, instead of maintaining a protective coating on the gear teeth, are forced out under even slight pressures to the sides of the pan, allowing the gear to cut a groove in which it rotates, the gear and pinion receiving no protection whatever. It is the practice of some traction companies to put large quantities of these greases in their gear pans, and the inexperienced inspector is often deceived by its action. The presence of a large quantity of grease on the rims, spokes, and even on the sides of the pan, leads him to believe that he is obtaining good lubrication, when, as a matter of fact, close inspection will usually show that the wearing surfaces of the teeth are absolutely dry and cutting. The failure of these greases to adhere to the teeth results in unnecessary wear, a shortening of the life of the gears, and makes a noisy traveling car, which excites comment on the part of the public.

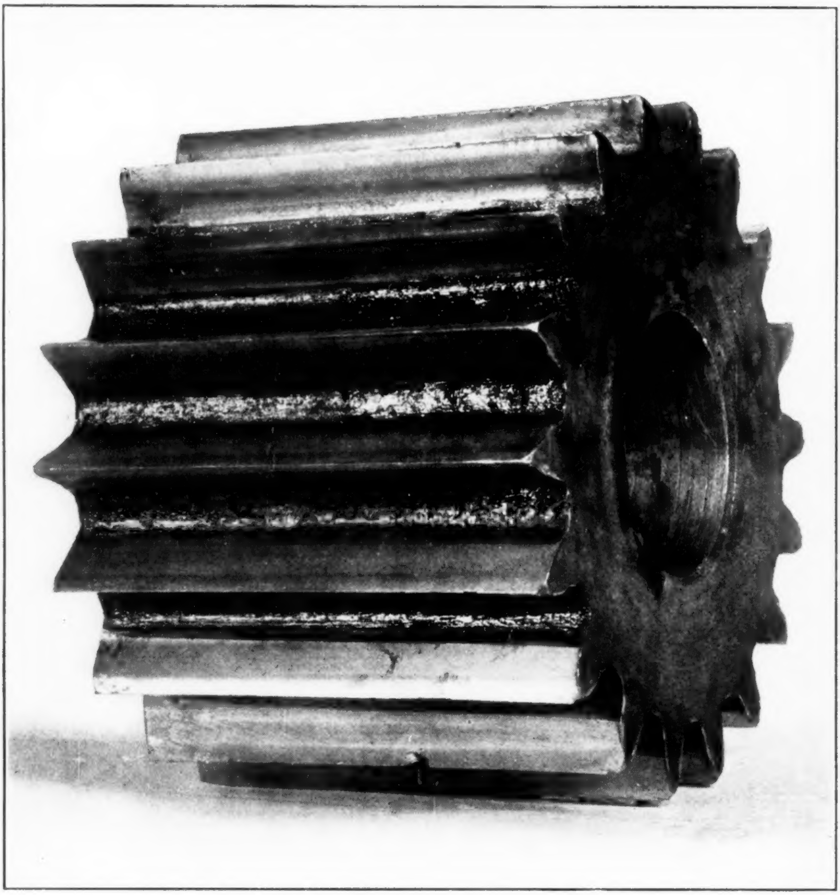


Figure 1—Pinion Teeth Worn to Knife Edge

Of peculiar interest is the performance of these greases in the presence of sand, grit, etc. These foreign substances added to the non-lubricating fillers employed in making gear grease make a perfect abrasive compound that actually produces wear on the gear and pinion teeth instead of preventing it. Fig. 1 shows the pinion taken from a car operated by a Westinghouse motor. The teeth are ground down to a sharp knife edge. Fig. 2 is a photograph showing four test tubes, one of which contains a sample of the new gear grease used to "lubricate" the gear. Beside it is shown the "filler" put in the grease to give it body. This has been removed from the grease by the use of 86° Baumé gasoline. The second sample shows the same amount of used grease, which was taken from the gear case, and beside it is the deposit which, in addition to the filler, contains sand, dirt, and iron. This foreign abrasive material does not settle out in a heavy grease, but is carried around with every particle of the grease.

Operating men who have studied the problem have realized the failure

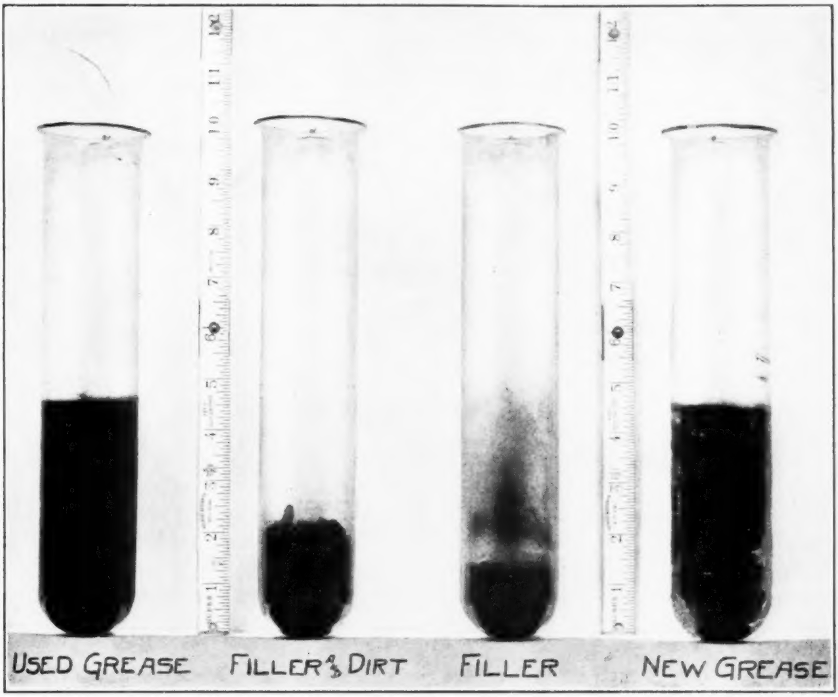


Figure 2
Used and Unused Grease Showing Insoluble Material

of the greases generally offered and are now demanding a heavier and more adhesive gear lubricant for their gears. This demand has become more urgent with the increased size of the cars, larger motors, and higher speeds.

In Crater Compound The Texas Company has produced an ideal lubricant which has been found to adequately meet all the exacting requirements of this class of work, and the results obtained have been very satisfactory to those who have used it. Crater Compound is an exceedingly heavy, straight mineral oil. It is a pure, homogeneous lubricant, free from any fillers. Its great adhesiveness enables it to resist the great pressure to which it is subjected in street railway gear lubrication, and to maintain the all-important film between the metal-bearing surfaces.

Another advantage of Crater is its durability. Where greases separate, dry up, and lose weight from service, Crater cannot be destroyed. After a short period greases become lighter in color, while Crater will retain a jet black color indefinitely. Crater is also of such consistency that it flows to the bottom of the pan, allowing the lower teeth of the gears to dip into it. Also, sand or road dirt drops through the Crater and settles on the bottom of the pan.

The writer recently conducted a test on a car of a large street railway, using Crater Compound in one gear case and grease in the other. The results of this test are tabulated below and show the relative amount of Crater and grease used.

Date	Miles	Grease No. 2 End Amount Applied	Grease No. 2 End Amount Consumed	Crater Compound No. 1 End Amount Applied	Crater Compound No. 1 End Amount Consumed
12/20/16	0	12 lbs.	....	$2\frac{3}{4}$ lbs.	....
1/5/17	944 (note)	11 lbs.	9 lbs.	$5\frac{1}{4}$ lbs.	$2\frac{1}{4}$ lbs.
1/26/17	984	5 lbs.	$9\frac{1}{2}$ lbs.	....	$3\frac{1}{4}$ lbs.
2/14/17	1,267	7 lbs.	5 lbs.	2 lbs.	$\frac{1}{2}$ lbs.
3/12/17	1,702	5 lbs.	$5\frac{1}{2}$ lbs.	....	$1\frac{1}{2}$ lbs.
4/5/17	1,320	10 lbs.	$6\frac{1}{2}$ lbs.	3 lbs.	1 lb.
Totals	6,217 miles	50 lbs.	$35\frac{1}{2}$ lbs.	13 lbs.	$8\frac{1}{2}$ lbs.

NOTE: Pinion teeth of No. 2 End badly chipped and 2 lbs. of grease removed because of slivers and broken chips in it.

From the table it is seen that for the same mileage, $8\frac{1}{2}$ lbs. of Crater Compound were consumed against $35\frac{1}{2}$ lbs. of grease. Measurements were also taken of pinion teeth, which showed that the wear on the gears lubricated with grease was .009 inches greater than on the gears lubricated with Crater. It will be noted that of five inspections made on the above car it was necessary to add more grease at each, while Crater was added at only three inspections, which would tend to show that gears lubricated with Crater Compound require fewer inspections and applications.

It will also be noted from the above that the amount of Crater applied in the gear case was considerably less than the amount of grease applied. Where grease is used it is the general practice of most companies to put large quantities in the gear pans, as much as 15 or 20 pounds per pan in many instances. With Crater Compound the practice is quite different. Its consistency, being of a soft nature, allows it to level back in the pan after coating all surfaces, and it is, therefore, necessary to apply only sufficient to allow the teeth to pick it up, which assures perfect lubrication at all times.

The quantity required for initial application will, of course, vary with the size of the gears and the clearance between the gear and the bottom of the pan. For small motors it has been found that three or four pounds per gear is sufficient, and for the larger type, as used on electric locomotives, this is increased to six or eight pounds. These quantities will allow the teeth to be immersed in a bath after the sides of the case and the gear surfaces have received a coating. For later applications about two pounds will be sufficient.

The accompanying diagram shows the results obtained with an initial amount of two pounds applied per gear case, and may be used as a guide in making applications.

In changing from gear grease to Crater Compound it is advisable that the grease be cleaned from the gears and pan in order to get the best results. This is absolutely necessary in making a test in order to allow Crater to show the best results, as the presence of grease or oil will interfere with the adhesive qualities of Crater. In making a test it is advisable, wherever possible, to apply Crater on new gears and pinions, using Crater in one gear case and the regular grease in another of the same car. Additional applications of each should be made as determined by inspection. In conducting such a comparative test, it is suggested that the initial amount of grease applied be the same as ordinarily used by the company, and subsequent applications

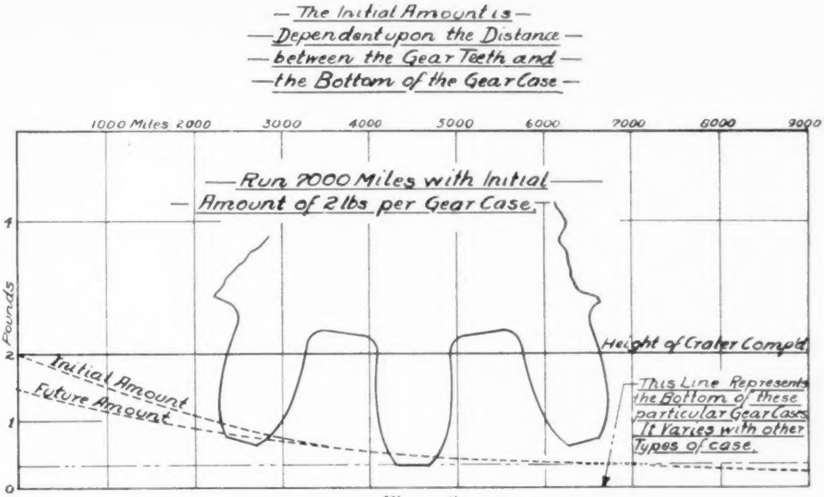


Figure 3

Diagram Showing Results of Test on Crater Compound

should conform with their regular practice. Mileage records should, of course, be kept and the consumption of each lubricant noted, as well as the condition of the teeth at each inspection. If possible, the gear and pinion teeth measurements should be taken in order to show comparative wear. This, in fact, is important, for it will show the saving in gear and pinion wear, and, consequently, in cost of maintenance, which is accomplished with the use of Crater Compound.

Crater is now being used on a number of railways, and reports from them are most satisfactory. The writer recently visited a large railway company, operating 1,200 cars daily, which has been using Crater exclusively in its gear cases for the past two years. The superintendent informed me that they averaged 2,333 miles per pound of Crater per gear. They had formerly used five or six times this amount of grease. He read an extract from his annual report in which he stated that his gear lubrication for the year showed a saving of more than $1,000.00 over the previous year. He also called attention to the fact that his pinion purchases had decreased from 780 for the previous year to 476, and gear purchases from 278 to 156, and stated that this saving was effected solely by the use of Crater Compound. He recently received a letter from the manufacturing company's engineer who furnishes him with gears and pinions, asking if he had ceased operating, as the customary orders were not coming through for gears and pinions. He replied that he had not stopped but was operating with a greater mileage than ever, and said that the sole reason for not ordering more gears and pinions was Crater Compound, which had reduced the gear wear to a minimum and had kept the cars out of the shop. He also informed me that measurements taken by the General Electric Company's engineer showed an average wear for gears and pinions of .005 inches for the year, which is unusually small.

Consideration of the many factors entering into the problem of efficient lubrication for this class of work, and of the qualifications of

Crater Compound to meet all the conditions, will show that Crater is far superior to all other forms of lubricants that may be used. Its efficiency not only insures decreased cost of lubrication, but also saving in gear and pinion wear, cost of maintenance, and keeps the car out of the shop. Crater is not affected by climatic conditions, giving the same service under high or low temperatures. In addition, its great cushioning effect prevents noise, even when the gear rubs against the pan. It will give remarkable results in preventing breakage of gear teeth under the most severe shocks, thus securing for the operator not only a saving in dollars and cents but insurance against noisy traveling cars, which is important if the good repute of the company is to be maintained.

A TEST ON TEXACO URSA OIL

By S. F. Lentz
Lubrication Engineer, The Texas Company, Chicago

A COMPARATIVE test of Texaco Ursa Oil and a competitive oil was recently made on a large pneumatic tool company's "Little Giant" horizontal oil engine. This engine is a 50-horsepower semi-Diesel two-cycle, single cylinder type, with 13-inch bore and 13-inch stroke. The engine is of the wet type, water being admitted to the combustion chamber separate from the fuel nozzle. The fuel is forced into the combustion chamber through a nozzle by a pump connected directly to the governor.

The engine cylinder is lubricated by one sight feed lubricator located on top of the cylinder. The main bearings, crank and cross head bearings are lubricated by splash feed from the crank case. During the run with the competitive products a different oil was used in the crank case from that in the cylinder, although the former had been recommended as, and purchased for, a cylinder oil. Texaco Ursa Oil was used in both the cylinder and the crank-case.

Considerable trouble had been experienced with this engine while using the competitive oil, which was a high priced lubricant of paraffine base and medium viscosity. The difficulty was that the engine, piston, and cylinder walls were covered with a very tenacious gum, which, after the engine had stood idle for a few hours, would prevent the piston from being moved and, consequently, the engine from being started, except by means of considerable labor and time in removing the side plate and applying a torch until the gum melted.

Owing to the difficulty found in removing the cylinder head and the time this would have taken, the lubricating film in the combustion end of the cylinder was not noted until after using the Texaco Ursa Oil, the plant manager stating that he was very familiar with the gummy condition of the cylinder walls and pistons with the competitive oil in use, as he had been obliged to remove the head a number of times for inspection. The condition at the front end of the cylinder, or the scavenging air compression end, was noted several times during each of the tests. The oil film on the cylinder walls at the front end of the cylinder was ascertained by the use of Riz-La-Croix cigarette paper. Throughout the tests general con-

ditions were kept as nearly identical as possible.

The following tables will show some of the results of the tests:

CYLINDER LUBRICATION

Lubricants	Length of Run—Hours	Average Drops per Min.	Total Gal. Fed to Cylinder	Rate per 10 Hrs. in Pints	Condition of Cylinder at End of Run
Competitive Oil—					
1st run	13.5	35	.19	1.13	Very dry; no sign of lubrication
2nd run	9.5	57	.23	1.9	Not so dry, but gummy deposit
TEXACO URSA OIL—					
1st run	7.5	40	.12	1.28	Good; some sign of gum from other oil
2nd run	6.75	38	.1	1.2	Excellent

LUBRICANTS IN CRANK-CASE

	Amount at Start	Hours Run	Amount at Stop	Total Loss	Percent. Loss	Condition at End of Run
Competitive Oil	2.43 gal.	23	1.8 gal.	.61 gal.	25%	Badly discolored
TEXACO URSA OIL	2.62 gal.	14	2.51 gal.	.11 gal.	4.2%	Practically uncolored

LUBRICATION COSTS—	*Cylinder Lubrication*		*Crank-case Lubrication*	
	Competitive Product	URSA OIL	Competitive Product	URSA OIL
Total time	23 hrs.	14.25 hrs.	23 hrs.	14.25 hrs.
Total consumed	.42 gal.	.22 gal.	.61 gal.	.11 gal.
Average per hour	.0183 gal.	.0154 gal.	.0265	.008
Cost per gallon	.60	.50	.45	.50
Cost per hour	.011	.0077	.012	.004
Cost per month (26 days)	$ 2.86	$ 2.00	$ 3.12	$ 1.04
Cost per 12 months	$34.32	$24.00	$37.44	$12.48
Saving per year by use of TEXACO URSA OIL		$10.32 or *30 per cent.*		$24.96 or *67 per cent.*

COMPARATIVE OPERATING COST OF LUBRICATING OIL

	Cylinder per Hour	Crank-case per Hour	Total per Hour	Total per 10 Hours	Saving per 10 Hours	Saving per 12 Months
Competitive Oil	.011	.012	.023	.23		
TEXACO URSA OIL	.0077	.004	.0117	.117	.113 or *49 per cent.*	$35.28

At the end of the first run on the competitive oil, no film at all was found; at the end of the second run the film was found to be of a black and gritty nature. This deposit would eventually cause the piston to stick when the engine had stood idle for a few hours. It should be mentioned that at the start of the test on the competitive oil the engine had been installed and operated only about two hours, so that all parts were perfectly clean. The piston and cylinder were not cleaned before starting the test on URSA OIL.

At the end of the first run on URSA the film, while light, was not decomposed and was of good lubricating value; at the end of the second run practically all of the gummy deposit had disappeared and much free oil was in evidence on the walls and piston. After the first run on URSA, though the engine had been standing idle in a zero temperature for some forty hours, the cylinder walls at the combustion end were covered with a clear, lubricating film, extending to the limits of the stroke each way. The man-

ager stated that with every other oil that he had tried, while feeding from two to three times as much as of the Ursa, the cylinder had shown a dry, burnt surface for the first two to four inches, with numerous dry spots elsewhere.

That URSA oil was effecting a much better seal between the cylinder walls and piston rings was evidenced by the difference in quantity and quality of the residue in the scavenging air compression chamber. With the competitive oil, this residue, aside from the water gathering there, amounted to over a pint per ten hours of engine operation, being of a black, burnt and gritty nature, with no sign of lubricating value, which proves that the greater part of the residue was fuel oil which had been permitted to pass the piston rings. With URSA oil the amount of the residue was so small that, during the entire run, it was impossible to obtain enough to fill an eight ounce bottle, and it was greasy, with practically no sign of fuel oil or carbon.

Another point which clearly showed the excellent effects of URSA in the cylinder was brought out by the manager's remark that the engine was running more easily than he had ever known it to run. The truth of this remark was proven by the fact that after about an hour's operation with URSA the engine began to pick up in speed and throughout the remainder of the test operated at from six to eight revolutions faster than with the competitive oil, both under the same load.

It was intended to run a further test with the competitive oil, but the manager did not want to take the chance of returning to the old condition by any further use of the old oil.

TECHNICAL REPORT

Hoffman & Curtis Bros.,
Ivanhoe, Minn.

Subject: Avery Tractor
Lubrication with Ursa Oil.

The above firm has customer to whom they sold a 40–80 h.p. Avery Tractor. The owner used 33 gallons of Ursa Oil in 36 days, averaging 15 hours a day. Previous to using Ursa Oil he used a well-known high-priced Motor Oil, using 64 gallons at 60 cents per gallon in 21 days, making a total cost of $38.40. He paid 60 cents per gallon for Ursa, at retail, making the total cost of his lubrication for 36 days, $19.80. He recently took down his motor on general principles, and no bearing showed wear, there was no carbon deposit, nor was there a loose bearing on the entire motor.

Comparative Tractor Operation and Cost Data

Make of Tractor—AVERY.

Size: 40-80 h.p.	Competitive Oil	Texaco Ursa
Total days	21	36
Hours per day	15	16
Total hours	315	510
Cost of oil per gallon	$0.60	$0.60
Total gallons used	64	33
Total cost	$38.40	$19.80
Gallons per day	3.04	0.91
Cost per day	$1.82	$0.54
Cost per hour	$0.12	$0.038
Decreased cost per day on Ursa	......	$1.27 or 70%
Decreased gallons per day on Ursa	......	2.13 gal. or 70%
Increased hours operation on Ursa	......	194 or 62%
Gallons used per hour	0.202	0.06
Decreased gallons per hour on Ursa	......	0.142 gal. or 70%
Condition of motor	Unsatisfactory	Very good

Dated 12/13/16.

(Signed) F. A. Rademacher.